A STUDY ON THE PERFORMANCE OF ALTERNATIVE REFRIGERANT MIXTURES FOR HCFC-22

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ABSTRACT

Performance of ternary mixtures composed of HFCs and HCs was researched to develop the HCFC-22 alternative refrigerant mixtures. We selected HFC-32, HFC-125, HFC-134a, HFC-143a, HFC-152a, HFC-227ea, and HFC-236ea as HFCs, and propane and isobutane as HCs. The simulator that can predict theoretically the performance of given refrigerant mixtures has been developed and tested for various refrigerant systems. Nineteen different kinds of ternary mixtures have been chosen for thermodynamic simulation. Among nineteen mixtures, six ternary refrigerant mixtures were selected as candidates for HCFC-22 alternatives. They were R-32/143a/600, R-32/152a/227ea, R-32/134a/236ea, R-32/143a/236ea, R-32/152a/236ea, R-32/134a/600a. Performance of these mixtures has been obtained experimentally by the thermodynamic calorimeter and was compared with that of HCFC-22, R-407C, and R-410A.

KEY WORDS: HFCF-22 alternative refrigerant mixtures, HFCs, HCs, performance, HCFC-22, R-407C, R-410A, simulator, calorimeter

1. INTRODUCTION

For nearly sixty years, chlorofluorocarbons (CFCs), have been widely used as solvents, foam blowing agents, aerosols and specially refrigerants due to their preeminent properties such as stability, non-toxicity, non-flammability, good thermodynamic properties and so on. However, they also have harmful effect on the Earth's protective ozone layer. So, they have been being regulated internationally by Montreal Protocol since 1989. Subsequently, it was discovered that CFCs also contributed significantly to the global warming problem. The result was that CFCs have been forbidden in developed from January of 1996. In 2010, producing and using of CFCs will be prohibited completely in all over the world. In consequence, lots of research have been done to find the suitable replacement for CFCs. Initial alternatives included some hydrochloro-flourocarbons, or HCFCs, but they will be also phased out internationally around 2020 ~ 2030 because their ozone depletion potentials and global warming potentials are in relative high levels though less than those of CFCs. Transitional compounds, such as HCFCs (hydrochloro-fluorocarbons), which are less harmful to the ozone layer, are to be used in their place until the year 2020. By that time compounds such as HFCs (hydrofluorocarbons), which are benign to the ozone layer, are expected to have replaced HCFCs. As a result, it became a very urgent issue to search and develop CFC and HCFC alternatives. HCFC-22 has been widely used as refrigerant in the room air conditioner. While HFC-134a was found to be a very ideal alternative refrigerant of CFC-12, HCFC-22 alternative has not been unfortunately found yet. It is generally accepted that the only choice to replace HCFC-22 is to mix two or more refrigerants together so that the mixture can bear the similar performance to HCFC-22. In general, COP, VCR, the pressure of evaporator, and the pressure of condenser determine the performance of refrigerant. In this work, we selected HFCs (HFC-32, HFC-125, HFC-134a, HFC-143a, HFC-152a, HFC-227ea, and HFC-236ea) and HCs (propane and isobutane) as HCFC-22 alternative refrigerants and simulated performance evaluation of refrigerant mixtures composed of HFCs and HCs for determination of final HCFC-22 alternative candidate mixtures. We compared R407C and R410A as well as HCFC-22 with refrigerant mixtures in COP, VCR, and capacity and after than, suggested several refrigerant mixtures that are available as HCFC-22 alternatives[1-2].

2. Simulation

2.1. Simulation of refrigeration cycle

Mechanical refrigeration systems are based on the principle that heat absorbed by a working fluid (refrigerant) when it changes from liquid to gas lowers the temperature of the objects around it. In the compression system, which is employed in electric home refrigerators and commercial installations, a compressor, controlled by a thermostat, exerts pressure on a vaporized refrigerant, forcing it to pass through a condenser, where it loses heat and liquefies. It then moves through the coils of the refrigeration compartment. There it vaporizes, drawing heat from whatever is in the compartment. The refrigerant then passes back to the compressor, and the cycle is repeated. The refrigerator is composed of compressor, condenser, and thermostatic expansion valve, and evaporator as four basic installations, and compressor fan, evaporator fan, liquid receiver, accumulator, suction line, liquid line in addition to these four basic installations. In refrigerant cycles, the P-h diagram was frequently used in the analysis of vapor-compression, and shown in Figure 1. In this diagram, three of the four processes appear as straight lines, and the heat transfer in the condenser and the evaporator is proportional to the lengths of the corresponding processes. In Figure 1, 1sp-2 process is compression, 2-4sb process is condensation, 4sb-5 process is expansion, and 5-1sp process is evaporation. In this simulation, four processes such as compression, condensation, expansion, and evaporation were modeled and simulated. The compressor was modeled on the assumption that it is compressed isentropically.

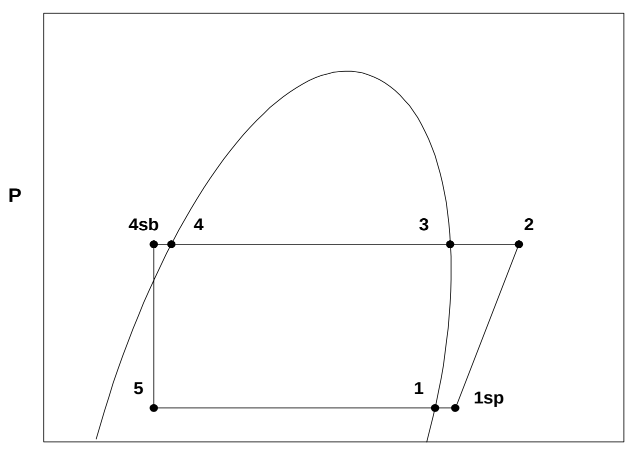


Figure 1. Pressure-Enthalpy Diagram

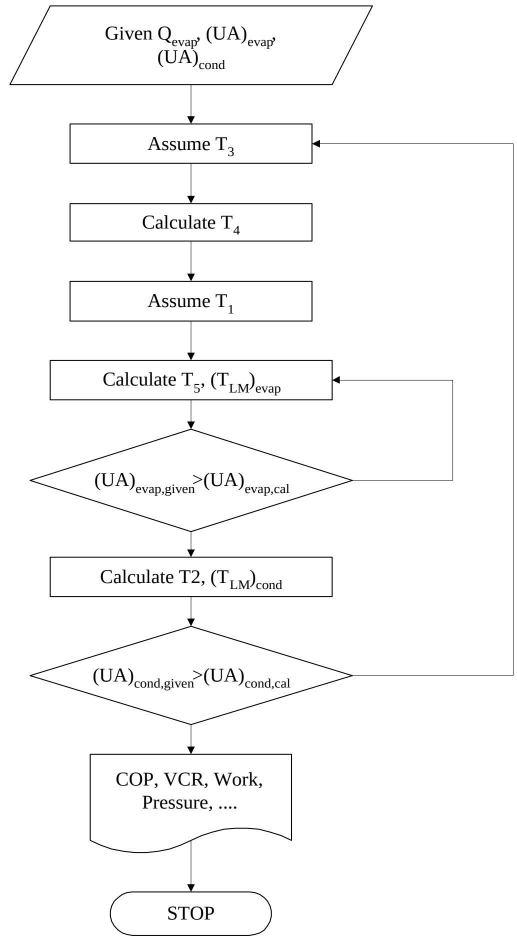


Figure 2. Flow chart for the thermodynamic cycle simulation

$$W_{comp} = \dot{m}(i_2 - i_{sp}) = \frac{\dot{m}}{\eta_{isen}}(i_{2,s} - i_{1sp}) \tag{1}$$

η_{isen} is isentropic efficiency during compression process, $i_{2,s}$ is enthalpy of state 2. We assumed that evaporator and condenser have no pressure loss and heat loss, and UA of evaporator and condenser is constant.

$$\mathrm{Q} = \mathrm{UA}\Delta\mathrm{T}_{\mathrm{LMTD}} \tag{2}$$

$$\Delta\mathrm{T}_{\mathrm{LMTD}} = \frac{(\mathrm{T}_{r2} - \mathrm{T}_{c2}) - (\mathrm{T}_{r1} - \mathrm{T}_{c1})}{\ln[(\mathrm{T}_{r2} - \mathrm{T}_{c2})/(\mathrm{T}_{r1} - \mathrm{T}_{c1})]} \tag{3}$$

r is refrigerant, c is second fluid, and expansion valve process was modeled isenthalpically.

$$\mathrm{i}_{4sb} = \mathrm{i}_5 \tag{4}$$

It ws supposed that there are no heat loss and pressure drop in every system. In refrigeration system, the representative performance characteristics were COP (Coefficient of Performance) and refrigeration capacity, and they are expressed as follows.

$$Q = \dot{m}(h_{1sp} - h_5) \tag{5}$$

$$COP = \frac{h_{1sp} - h_5}{h_2 - h_{1sp}} \tag{6}$$

The experimental conditions were listed in Table 1 and flow chart for the thermodynamic of cycle simulation was shown in Figure 2.

2.2. Selection of ternary refrigerant mixtures

Performance of nineteen ternary refrigerant mixtures composed of HFCs and HCs for HCFC-22 alternatives were evaluated by simulation program called 'Simcycle'. Nineteen mixtures that were not published yet as HCFC-22 alternatives were selected. These mixtures were listed in Table 2. The simulation results such as COP, VCR, P_L, and P_H for nineteen ternary mixtures were compared with those of HCFC-22, R407C, and R410A listed in Table 3 [3-5].

2.3. Results of simulation

Among nineteen mixtures, six ternary mixtures having good performance comparing with HCFC-22 were selected by the thermodynamic cycle simulation evaluation as HCFC-22 alternatives. These refrigerant mixtures and compositions were listed in Table 4. These six selected ternary mixtures were non-azeotropic mixtures and the temperature gradient of those mixtures was 2.17~8.27^{o}C at 101.325 kPa. According to the simulation results, COP of ternary mixtures except for R-32/143a/236ea (80/10/10wt%) and R-32/134a/227ea

Table 1. Experimental condition for performance test of refrigerant mixtures

Computer simulation	
Compressor input T(^{o}C) of second fluid	25 ^{o}C
Compressor output T(^{o}C) of second fluid	35 ^{o}C
Evaporator input T(^{o}C) of second fluid	15 ^{o}C
Evaporator output T(^{o}C) of second fluid	5 ^{o}C
Capacity of air-conditioning	2 kW
Total heat flow rate of evaporator (UA)	0.20kW/K
Total heat flow rate of evaporator (UA)	0.24kW/K
Degree of super-cooling from compressor	5 ^{o}C
Degree of super-heat from evaporator	5 ^{o}C
Efficiency of compressor	0.8
Experiment	
Saturated T(^{o}C) of induction pressure	7.2 ^{o}C
Induction gas T(^{o}C)	35 ^{o}C
Saturated T(^{o}C) of nozzle pressure	54.4 ^{o}C
Degree of supercooling of liquid-refrigerant	8.3 ^{o}C
Surrounding T(^{o}C) of compressor	35 ^{o}C

Table 2. Ternary mixtures evaluated by the thermodynamic cycle simulation

	Mixtures	Compositions
1	R-32/143a/C270	20-40wt%/30-70wt%/0-30wt%
2	R-32/143a/227ea	30-70wt%/10-70wt%/0-45wt%
3	R-32/143a/245cb	30-60wt%/20-70wt%/0-30wt%
4	R-32/143a/236ea	30-60wt%/20-70wt%/0-20wt%
5	R-32/143a/600	30-50wt%/38-70wt%/0-12wt%
6	R-32/143a/E134	30-60wt%/32-70wt%/0-8wt%
7	R-32/143a/E245	30-70wt%/20-70wt%/0-10wt%
8	R-32/218/152a	40-70wt%/0-40wt%/10-40wt%
9	R-32/C270/245cb	50-70wt%/0-28wt%/20-40wt%
10	R-32/134a/227ea	35-70wt%/0-60wt%/5-30wt%
11	R-32/134a/600a	30-70wt%/20-65wt%/0-20wt%
12	R-32/134a/236ea	35-70wt%/10-60wt%/0-20wt%
13	R-32/152a/227ea	40-70wt%/0-50wt%/0-40wt%
14	R-32/152a/600a	45-70wt%/5-50wt%/0-25wt%
15	R-32/152a/236ea	45-70wt%/0-50wt%/0-20wt%
16	R-32/152a/600	45-80wt%/10-45wt%/0-20wt%
17	R-32/152a/E134	60-80wt%/12-35wt%/0-8wt%
18	R-32/152a/E245	50-70wt%/20-40wt%/0-10wt%
19	R-32/600/E134	80-96wt%/0-12wt%/4-10wt%

(80/10/10wt%) were smaller than that of HCFC-22. In case of R-32/143a/236ea (80/10/10wt%), COP was better than that of HCFC-22 but the input-output pressures of compressor were higher than that of HCFC-22. VCR of R-32/152a/227ea (40/40/20 wt%) is similar to HCFC-22 and VCR of others are higher than that of HCFC-22. The pressure ratio of R-32/143a/236ea (80/10/10wt%) was the smallest in all mixtures, and 11% smaller than that of HCFC-22. The pressure ratio of R-32/134a/227ea (30/50/20wt%) was the largest, and 15.5% larger than that of HCFC-22. The input-output pressures of all mixtures of compressor were higher than that of HCFC-22. If the input-output pressures of compressor are high, all parts and accessories of instruments should be replaced for high pressure. As a result, R-32/143a/236ea (80/10/10 wt%) and R-32/152a/227ea (40/40/20 wt%) are considered to be able to alternate HCFC-22 in the viewpoint of performance and having similar properties to HCFC-22, respectively

Table 3. Comparison of refrigeration performance of HCFC-22, R407C, and R410A

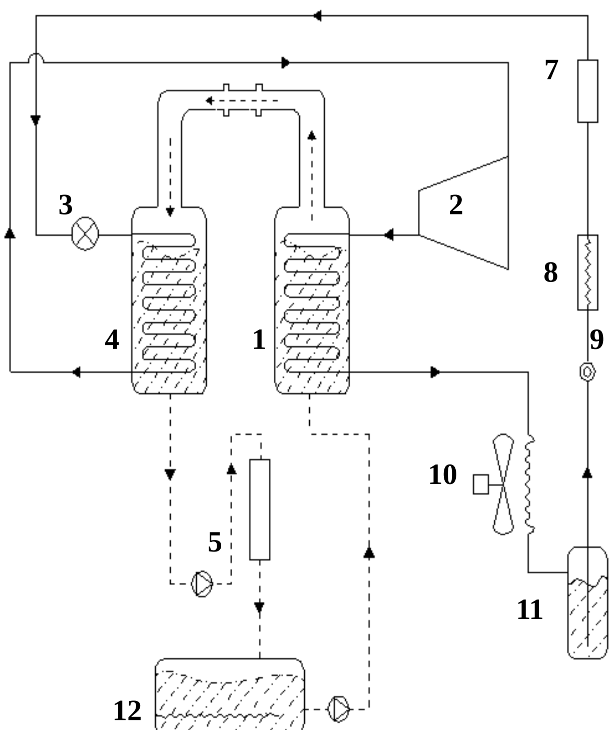
	Compositions (wt%)	COP	VCR (kJ/m^3)	P_L	P_H
HCFC-22	**HCFC-22 = 100**	**5.45**	**3338**	**455**	**1254**
R407C	HFC-32/125/134a = 23/25/52	4.98	3412	460	1445
R410A	HFC-32/125 = 50/50	5.31	5117	730	1993

Table 4. Six ternary mixtures and their compositions selected by simulation

NO.	Ternary mixtures	Compositions
1	HCFC-22	100wt%
2	R410A (HFC-32/125)	50 wt%/50 wt%
3	R407C (HFC-32/125/134a)	23wt%/25wt%/52wt%
4	**HFC-32/143a/600**	**30-50wt%/ 38-70wt%/ 0-12wt%**
5	**HFC-32/152a/227ea**	**40-70wt%/ 0-50wt%/ 0-40wt%**
6	**HFC-32/134a/236ea**	**35-70wt%/ 10-60wt%/ 0-20wt%**
7	**HFC-32/143a/236ea**	**30-60wt%/ 20-70wt%/ 0-20wt%**
8	**HFC-32/152a/236ea**	**45-70wt%/ 0-50wt%/ 0-20wt%**
9	**HFC-32/134a/600a**	**30-70wt%/ 20-65wt%/ 0-20wt%**

3. Performance evaluation of alternative ternary refrigerant candidates

3.1. Calorimeter evaluation



1. Condenser **2.** Compressor **3.** Expansion valve **4.** Evaporator **5.** Pump 1 **6.** Pump 2 **7.** Flow meter **8.** Electric heater **9.** Sight glass **10.** Sub condenser **11.** Receiver **12.** Storage tank

Figure 3. Schematic diagram of refrigeration

In this part, the performance of six ternary refrigerant mixtures for HCFC-22 alternatives selected by the thermo-dynamic cycle simulation was evaluated experimentally by using calorimeter. The performance of candidate mixtures was evaluated by comparing their measured COP (Coefficient of Performance), Q_e (Refrigeration Capacity), and VCR (Volumetric Capacity of Refrigeration). These values were calculated by following equations.

$$COP = \frac{Q_e}{W_e} \qquad (7)$$

$$Q_e(W) = \dot{m} \times h_{fg} \qquad (8)$$

$$VCR(kJ/m_3) = \frac{h_{fg}}{V} \qquad (9)$$

$\dot{m}$ (g/s) is mass flow rate of first refrigerant, h_{fg} is enthalpy difference between inlet and outlet of evaporator, V_v (m^3/kg) is specific volume of compressor inlet, and W_e (W) is electric power consumption. In this work, the thermodynamic properties were obtained from REFPROP (V.6.01, NIST) [6]. Six mixtures selected by simulation were used for performance evaluation, and

Table 5. Compositions of mixtures for refrigeration performance ([a]R410A and [b]R407C)

	HFC-32	HFC-125	HFC-143a	HFC-22	HFC-134a	HFC-152a	HFC-227ea	HFC-236ea	R-600	R-600a
1				100						
2[a]	50	50								
3[b]	23	25			52					
4	56.3		39						4.7	
5	70					9.8	19.7			
6	65				30			5		
7	60		30					10		
8	54.7					40.5		4.8		
9	41.8				47.3					10.9

composition of those mixtures was listed in Table 5. The evaluation of refrigeration performance was carried out by calorimeter, and the experimental condition was listed in Table 1 and the schematic diagram of this apparatus was shown in Figure 3.

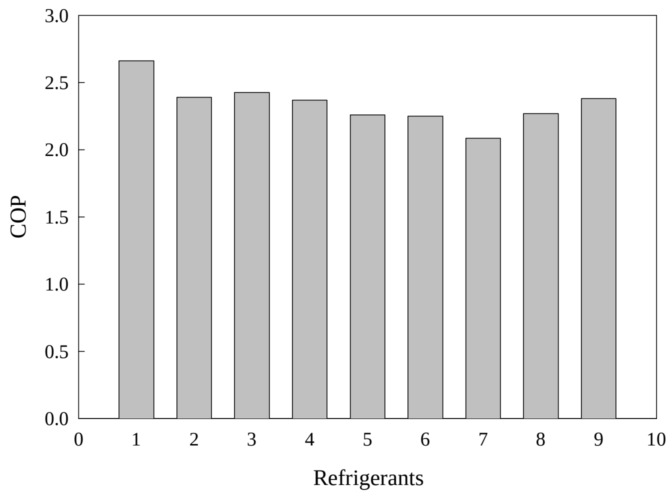
3.2. Results of calorimeter evaluation

The COP of ternary mixtures was shown in Figure 4. In this Figure, COP of all mixtures was 10.59~21.67 % smaller than that of HCFC-22 (COP=2.663). In addition, COP of R410A (COP=2.391), and R407C (COP=2.427) that are well known as HCFC-22 alternatives was smaller than that of HCFC-22 in this experimental apparatus. Among six candidate ternary mixtures, performances of R32/143a/600 (56.3/39/4.7wt%, COP=2.37) and R32/134a/600a (41.8/47.3/10.9 wt%, COP=2.381) were similar to R410A well known as HCFC-22 alternatives. Capacity of R407C and R32/152a/236ea (54.7/40.5/4.8wt%) was smaller than that of HCFC-22. Figure 5 shows the refrigeration capacity of ternary mixtures. Refrigeration capacity of every mixtures was 1.64～23.17% larger than that of HCFC-22 except for R32/152a/236ea(54.7/40.5/4.8wt%). Refrigeration capacity of R32/143a/600 (56.3/39/4.7wt%) was the largest among these mixtures and 23.17% larger than that of HCFC-22, and 54.64% larger than that of R410A. Figure 6 shows the VCR of mixtures. VCR of R410A in these mixtures was the largest, VCR of R32/152a/236ea (54.7/40.5/ 4.8wt%) and R32/134a/600a (41.8/47.3/10.9wt%) was similar to that of HCFC-22. In case that VCR of mixtures was similar to that of HCFC-22, the compressor for HCFC-22 was usable without development of new compressor. The experimental data of COP, refrigeration capacity, and VCR for HCFC-22, R410A, R407C and six candidate alternative refrigerant mixtures were listed in Table 6.

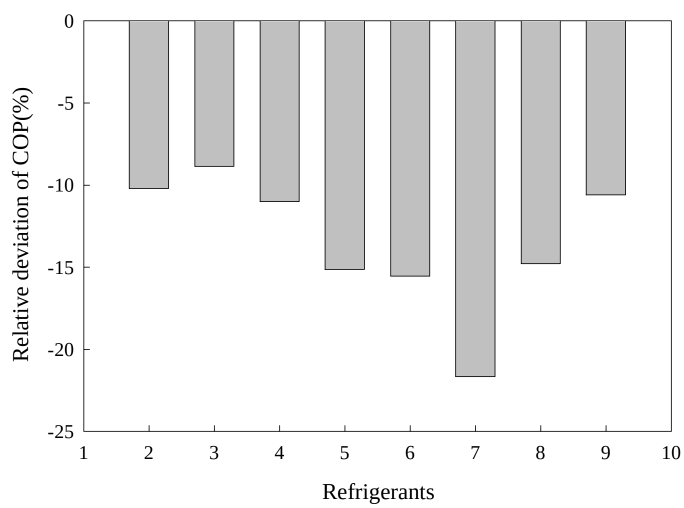
Table 6. Results of performance experimented by the calorimeter

	COP	Refrigerant capacity	VCR (kJ/m^3)	P_H (kPa)	P_L (kPa)
1	2.663	3012.1	3941.73	2147.245	623.055
2	2.391	4261.6	5940.39	3385.478	998.515
3	2.427	2863.9	3779.07	2439.934	587.796
4	2.370	3920.9	5311.55	3207.902	925.977
5	2.260	3485.0	4830.56	3070.427	778.361
6	2.249	3334.9	4626.03	2963.595	726.953
7	2.086	3412.6	4889.74	3200.392	817.114
8	2.269	2755.8	3739.87	2520.763	565.564

9	2.381	3062.4	4058.99	2665.191	666.500

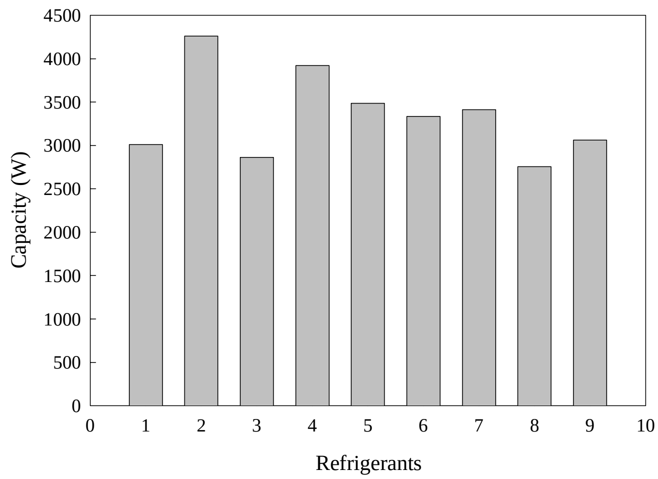


(a) COP

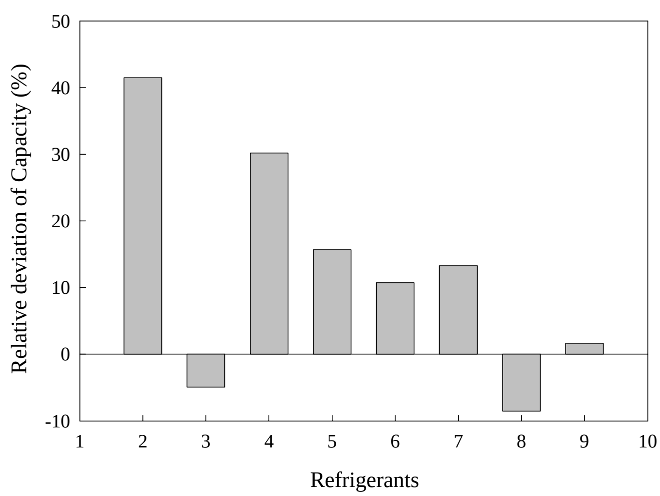


(b) Relative deviation of COP compared with HCFC-22

Figure 4. COP of ternary mixtures and relative deviation of COP compared with HCFC-22; 1- HCFC-22; 2-R407C; 3-R410A

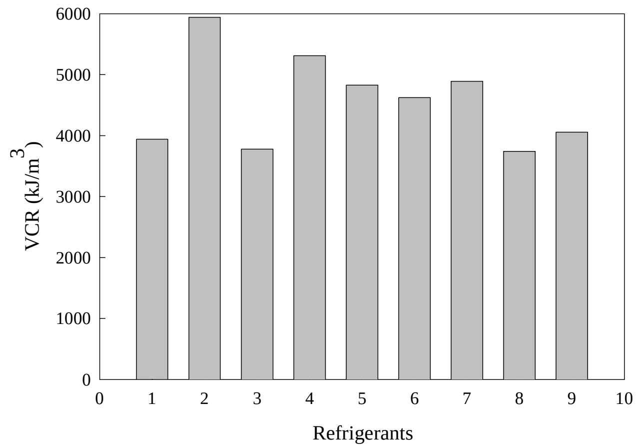


(a) Capacity

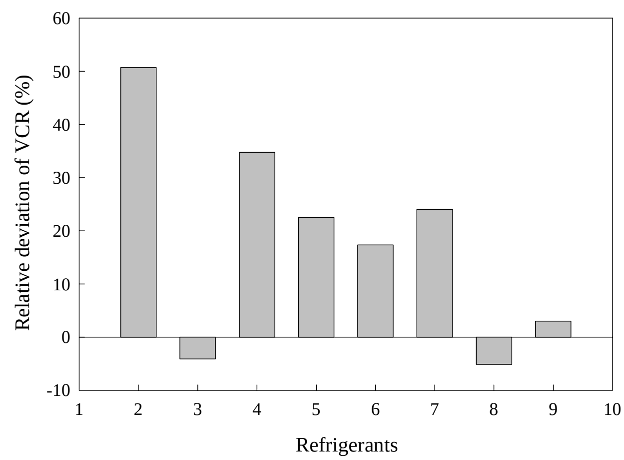


(b) Relative deviation of capacity compared with HCFC-22

Figure 5. Capacity of ternary mixtures and relative deviation of capacity compared with HCFC-22.



(a) VCR



(b) Relative deviation of VCR compared with HCFC-22

Figure 6. VCR of ternary mixtures and relative deviation of VCR compared with HCFC-22, 1-HCFC-22; 2-R407C; 3-R410A

4. CONCLUSION

Performance of ternary mixtures composed of HFCs and HCs was researched to develop the HCFC-22 alternative refrigerant mixtures. Six candidate refrigerant mixtures were selected out of nineteen ternary mixtures composed of HCs and HFCs by the performance evaluation using the thermodynamic simulation. Performance such as COP, refrigeration capacity, and VCR of mixtures selected was evaluated experimentally by calorimeter test and the characteristics of these mixtures were investigated. These results are expected to be useful as important technical information in the refrigerant industry.

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